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A PROCEDURE FOR ESTIMATING COSTS OF THERMAL EFFLUENT MODIFICATIONS FOR EXISTING STEAM-ELECTRIC GENERATING STATIONS

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May 1972

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A PROCEDURE FOR ESTIMATING COSTS OF THERMAL EFFLUENT MODIFICATIONS FOR EXISTING STEAM-ELECTRIC GENERATING STATIONS

Study Paper No. 2

for the

National Thermal Pollution Research Program

EPA, WQO Contract No. 14-12-837

An Economic Analysis of Thermal Pollution Abatement Costs in the Electric Power Industry

CEM Report No. 4059-464
May 1972

Leonard Ortolano
Frank Austin Smith

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INTRODUCTION

Because of important practical differences in engineering design options available and the nature of many of the economic cost tradeoffs involved, we have considered it manditory in analyzing thermal effluent modifications to draw a clear distinction between steam-electric power plants that are already in existence and those that are still in the design phase of development. The latter, "proposed plants," case has received the major part of our attention, and our cost estimating procedure and case study results for proposed plants are presented in our final report document. Since our procedure and results with respect to existing plants have not been sufficiently developed to a level warranting inclusion in a final report, the present study paper has been prepared as a supplementary progress status report on this aspect of the project.

Plants already in existence can themselves be distinguished by the nature of the cooling system employed. In regions where water available for cooling purposes is scarce, there has been a tendency to employ either cooling towers or ponds. These towers or ponds may be considered as part of a privately optimal plant design in the sense that their use is dictated by the absence of a relatively inexpensive water supply of adequate quantity for once-through cooling; this is all quite apart from considerations relating to thermal pollution. Although there are circumstances in which thermal pollution may be an issue for existing plants having towers or ponds, these situations are not common, and consequently we will not direct our attention to them.

The existing plants of concern herein are those with once-through cooling systems designed in the absence of explicit thermal effluent constraints.

This includes nearly all plants with once-through cooling systems constructed prior to the late 1960's, the period during which most states established numerical standards controlling the extent to which cooling system effluents could be discharged to natural water courses.

In the absence of thermal quality standards, the volume flow rate and temperature rise of condenser cooling water are determined largely by economic considerations revolving around the links between the cooling system, the condenser vacuum and the station heat rate. Within limits that depend on the turbine employed, it can be expected that increased flow of a given intake temperature will result in a lower condenser vacuum and a decreased station heat rate. Although the latter effect decreases fuel costs, the need to intake higher flows increases some capital and operating costs (e.g., larger pipes and increased pumping are required). It is this sort of economic trade-off analysis that establishes the extent of heated effluent discharge in the absence of explicit thermal quality constraints. Throughout, we shall refer to plants designed in the absence of thermal discharge constraints as "privately optimal designs."

The present paper analyzes the possible changes that could be made in such privately optimal designs to modify the heated effluents so that they conform with thermal quality standards; the latter are specified in terms of maximum allowable combinations of discharge flow rate and temperature. Thermal standards that can be met with the privately optimal design are of little interest, and consequently, we only examine situations in which standards cannot be met without some kind of modification. Actions taken to alter the cooling system effluent to conform with the standard are referred to as "thermal effluent modifications." The objective of the method of analysis presented below is the determination of that combination of effluent modifica-

tion options which meets the standard at minimum cost, subject to a constraint on total energy output.

The constraint on energy output is a consequence of the assumption that regardless of the effluent modifications employed, the output that would have been produced by the privately optimal plant in the absence of thermal quality constraints will be produced. Therefore, if we consider decreasing production as a way of modifying the cooling system effluent, then the cost of obtaining supplementary power from an alternative source must be included as part of the total cost of this modification.

We postulate that all effluent modification options that might be economically feasible can be divided into two classes, namely, reductions in energy output for the plant in question, and changes in the cooling system.*

The remainder of this study paper elaborates on the method of analysis developed and reports on progress made in making this method operational. Unfortunately, since we encountered several unanticipated difficulties we were unable to carry this portion of the work to a final state, and so we have no numerical results or illustrative case studies to report.

* We consider it very unlikely that a utility would consider redesigning the steam cycle to obtain a lower heat rate as a method for meeting a thermal quality constraint. However, this possibility may be feasible for proposed plants and has been examined elsewhere.

GENERAL DESCRIPTION OF THE PROPOSED METHOD OF ANALYSIS

The method of analysis essentially consists of examining alternative combinations of two types of thermal effluent modification options, namely, decreasing output and altering the cooling system. This section provides a descriptive overview of the method.

Production Decrease

The possibility of operating a unit of a power plant at a lower level than would be employed in the absence of thermal quality constraints constitutes an abatement option that is physically meaningful and may be economically sensible. The technological feasibility of this option is established by the fact that units are operated at less than full capacity, sometimes for extended periods of time, when the economics of load dispatching dictates that they should be. As an example, consider the so-called "swing-unit" in a utility system; i.e., a unit operating with very high variations in percent of capacity used over time to "pick up" the short term variations in demand on the system.

It is not at all clear, however, whether using a decrease in production to control waste heat will make economic sense. In general, this will depend on a host of circumstances which are local in the sense that they depend on the role which the unit plays in the overall system, as well as more obvious local features relating to fuel and labor costs.

In assessing the economic feasibility of this control option a central role is played by the assumption that no matter what the decrease in production, it will have to be made up by power from an alternative source. To reckon this cost it is necessary to set down what we refer to as the privately optimal use schedule, i.e., the manner in which the unit would have been operated in the absence of thermal quality constraints. It is also necessary to define the nature of the production decrease being considered; i.e., an alternative plant utilization schedule. Once these use schedules are postulated, it is possible

to determine the power output that must be obtained from an alternative source. Given the cost of power from an alternative source, it is possible to compute the cost of making up the deficit.

However, the cost of making up the deficit in power output is but one aspect of the total cost involved in employing a production decrease as an abatement option. This results because station heat rate and condenser duty both vary with the level of capacity used. Figures 1 and 2, which are based on data from a fossil fueled plant in the Northeastern United States, demonstrate the nature of these variations. As indicated in the figures, a decrease in the level of plant utilization leads to an increase in station heat rate and a less than proportionate decrease in condenser duty. The former change must be reflected in an increase in fuel cost for the power actually generated using the existing unit. The change in condenser duty must be accounted for in computations relating to cooling system modification which are taken up below.

The analysis of the implications of using a production decrease as an abatement option is not complete. It is still necessary to trace the implications of an alternative use schedule on the cooling system, and in particular, on the characteristics of the cooling system effluent. An alternative use schedule that does not, in and of itself, assure attainment of the standards, should still have utility in its effect on decreasing the extent of cooling system modifications necessary to meet the standards. This is so because the heat rejected to the cooling system will be lower than what it would have been if the privately optimal use schedule were employed.

The major complication in analyzing alternative plant utilization schedules as an abatement option results from the fact that plant utilization, climatic conditions and even the thermal quality standards themselves, can be expected to change during different periods within a year. Thus, it becomes necessary to consider production decreases in seasonal terms. The nature of this complication is conveniently presented with the aid of an example.

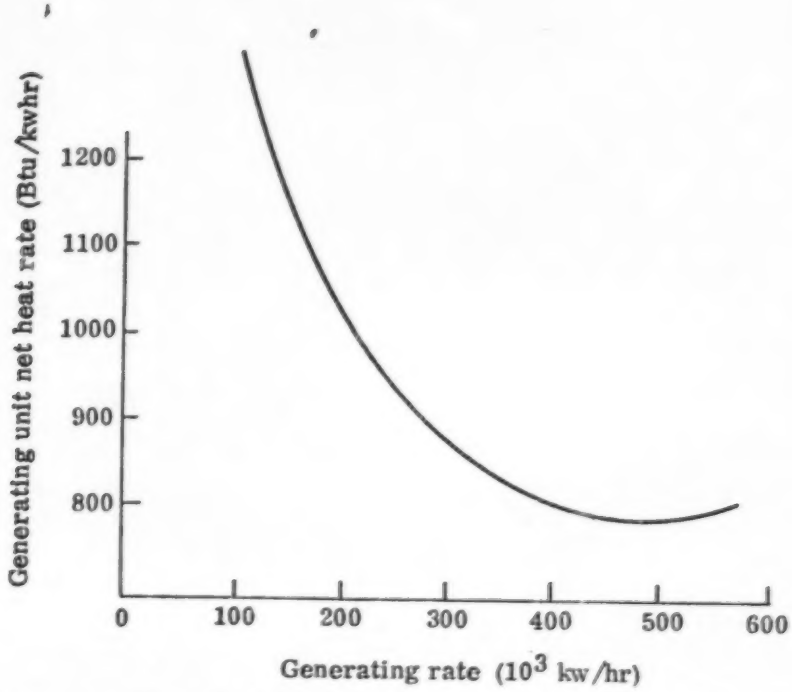

Figure 1. Unit net heat rate in relation to level of capacity utilization.

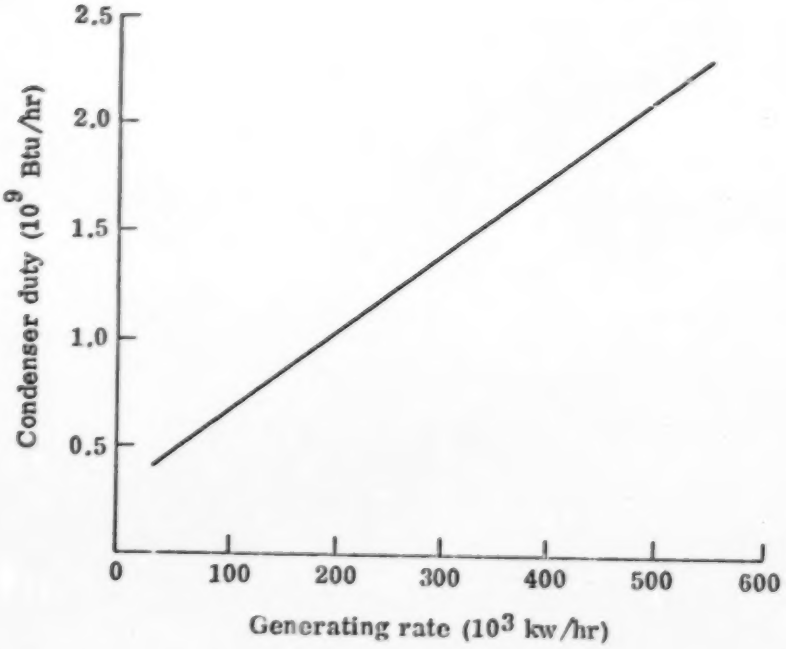

Figure 2. Condenser duty in relation to level of capacity.

Suppose the plant under consideration (using the privately optimal use schedule) was capable of meeting thermal quality standards in all periods except for summer. In this instance production decreases would be considered only for the summer and could be defined as follows: the unit will never be operated at more than α percent of full capacity in summer. Different values of α would correspond to different control options. For very low values of α it is likely that the thermal quality standards could be met without any change in the cooling system. For higher values of α it would be necessary to modify the cooling system (perhaps by adding a cooling tower), but these modifications need only be used during summer.

Now consider a case where employment of the privately optimal use schedule would lead to violations in the standards throughout the year. In this instance a production decrease control option would be specified in terms of the maximum levels of plant utilization for each season. Once these levels are specified, it is necessary to see if the thermal quality standards would be violated during any of the seasons. If there are no violations, then there is no need for cooling system modifications. However, if the standards are violated, then it is necessary to modify the cooling system in such a way that violations will not occur. Given the temporal variations in energy production, climatic variables and thermal quality standards, determining the "design conditions" for such a cooling system modification would require a trial and error procedure.

Cooling System Modification

The discussion above focused on the possibility of using decreases in production as a means for reducing the discharge of waste heat, and paid only casual attention to the implications of doing so on the requisite extent of cooling system modification. In this section we focus on the analysis of cooling system modifications.

The details of the procedure (i.e., assumptions, formulae, etc.) are contained in the Appendix. In general terms, the procedure determines cooling system modifications that transfer the design heat load at the condenser and meet the thermal quality constraints. There will be a large number of modifications that satisfy these conditions, and the procedure is designed to select from these the modification involving the lowest cost.

These alternative modifications are distinguished by type as follows:

- Once-through, with and without blending -- the modification of once-through cooling systems by altering the water flow rate through the condenser. The blending schemes consist of taking water (from the intake) directly to the cooling system outlet and mixing it with condenser outflow to decrease the effluent temperature.
- Recirculation -- the formation of a closed loop for circulating water flow, with the addition of a heat rejecting device (e.g., cooling tower or pond) to remove the heat added to the circulating water flow at the condenser.
- Helper -- the addition of a heat rejecting device to cool part or all of the outflow from the condenser prior to its discharge to a receiving water.

Within any one type, the modifications are distinguished by variations in important design variables, for example, the quantity of water flowing through the condenser.

The viewpoint adopted in deriving alternative designs is that a utility would make an effort to minimize the extent to which a unit would be out of operation as a result of cooling system modifications; and moreover, the utility would be inclined to use as much of the existing plant and equipment as possible,

consistent with the modifications to be made.* Consequently, we assume that the condenser(s) in the existing plant will not be replaced; i.e., the time period required in keeping the unit out of production while switching condensers would impose a prohibitive expense. Space availability in the original structure and the cost of physically tearing out a major component like a condenser are additional factors supporting the assumption. In the same spirit, we assume that the existing pipes and pumping facilities will be used to the extent possible.

A rather important consideration in determining the feasibility of making modifications to the existing cooling system is the expected remaining useful life of the unit. This enters into our approach by way of the fixed charge rate which, following a standard procedure employed in power plant economic analysis, is used to annualize capital expenditures. The shorter the remaining useful life of the plant, the higher the fixed charge rate applicable to current investment for thermal effluent modification.

There are three assumptions implicit in the method of analysis that require special mention. The first concerns the manner in which the capacities of the added component (i.e., pipes, pumps, etc.) are determined. In dealing with this we relied quite heavily on the notion of a "design condition"; i.e., a single set of steam cycle related variables (viz., condenser duty and vacuum), climatic variables, and thermal quality standards. In other words, the capacity of added components is determined on the basis of meeting a specified thermal quality standard under this simple set of conditions relating to the condenser and climate. The design conditions are selected such that the modifications sized on this basis will be capable of meeting the thermal quality standards during any period of the year.

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* In developing the assumptions and viewpoint for the existing plant situation we were influenced by the advice of LeRoy Newbold of Jackson and Moreland, Boston, Massachusetts.

In many cases, selection of design conditions will be reasonably straightforward. Typically the conditions would relate to an adverse climatic situation within the summer with the plant fully loaded. However, in cases where the production decrease option were employed to restrict output in summer, it might not be obvious as to which seasonal condition would be appropriate for component sizing; in such cases a trial and error procedure would be required.

The second assumption requiring special mention concerns the nature and approach to penalty costs associated with shifting station heat rates. We approached this issue by reasoning that under design conditions there would be a particular condenser duty (H*) and vacuum (p*) that are associated with the existing plant when it is operated at a postulated level of utilization. This level is specified in connection with the production decrease option discussed above. However, it would restrict the range of alternative cooling system designs unnecessarily if we required that the condenser duty and vacuum be maintained at these levels. This follows, since nearly all of the possible modifications will result in condenser vacuums different from those associated with the existing system.

Thus, a portion of the analysis must deal with the implications of what amounts to design values of condenser duty and vacuum different from H* and p*, respectively. In cases where the modification of the cooling system would yield design values greater (less) than p*, the station heat rate would increase (decrease) over what it would have been using the existing system. This in turn would influence both the fuel cost and the maximum possible output (i.e., capability) for the plant. Thus, changes in heat rate require that we apply a penalty (or benefit) to account for the change in fuel cost. Also, given the assumption that demand be met, changes in capability require that we account for the cost of maintaining the capability levels of the existing plant, as for example, by obtaining power from an alternative source.

While we recognized that these costs had to be accounted for, we tried to accomplish this in a computationally efficient manner. The standard approach involves accounting for the operation of the plant during different seasons and at various levels of loading throughout a typical year.* This is done for the existing system and for the existing plant with a modified cooling system. The appropriate penalty costs are determined on the basis of resulting differences in overall fuel costs and plant capability levels. This more or less standard approach is manageable in circumstances where only a relatively small number of alternative designs are involved, but we judged (perhaps inaccurately given what we learned) that it would be too burdensome computationally since we hoped to examine as many as several hundred alternative designs. Therefore, we abandoned this seasonal accounting approach as being unworkable for our purposes. Unfortunately, the approach we did follow (discussed below) turned out to have limitations that are so severe, that it seriously restricts the utility fo our approach in its current form.

Our earliest approach consisted of determining the shift in station heat rate under design conditions. Knowing this we computed the change in the maximum capability of the unit and applied a penalty cost for capability lost. For reasons that will be clarified below, we assumed that the fuel input to the plant with the modified cooling system would be the same as that for the plant with the existing cooling system. The cost penalty was then determined on the cost of additional fuel for power output needed, but not generated at the plant due to the shift in heat rate. We originally anticipated that these penalty costs would not be a major issue, but the preliminary computations we performed provided convincing evidence that we were wrong. Indeed, for existing plants, such penalties may be quite significant relative to other cost items, and our

* For an example of how this can be accomplished where a small number of alternatives is involved see Fuller, W.P., "A Survey an Economic Analysis of Alternative Methods for Cooling Condenser Discharge Water in Thermal Power Plants," Dynatech Report No. 921, Dynatech Corp., Cambridge, Mass., July 8, 1970.

simplistic approach is completely inadequate.

The third assumption requiring further elaboration concerns the use of a constant fuel input (i.e., firing rate) for the analysis of alternative cooling systems. The essential motivation for this assumption is the desire to permit analysis of production decrease options separately from (or rather sequentially with) cooling system modification options. To see this more clearly consider a situation where the production decrease option was defined in terms of setting the maximum level of plant utilization at 50 percent with the condenser vacuum at p^*. A plant operated to yield 50 percent of the full capacity output when the privately optimal condenser vacuum obtained, would yield less than 50 percent if the vacuum were greater than p^*; i.e., under these circumstances the heat rate would increase giving more waste heat per unit of fuel input.

To guarantee that the plant's output corresponded to that which obtained using 50 percent of capacity at the privately optional backpressure, we could modify the production decrease, i.e., assume a somewhat higher fuel input to compensate for the loss in efficiency. However, this procedure requires that the production decrease option cannot be even temporarily analyzed separately from the cooling system design. In fact a computationally messy iterative procedure would be required to guarantee that the output from the unit was at a particular value. We found it desirable to avoid this by the fixed fuel input assumption which essentially permits the temporary separation of the production decrease portion of the analysis from the cooling system design. In performing the analysis of the latter we assume that the fuel input to the boiler is fixed at the value corresponding to the production decrease decision. For example, specifying output levels at 50 percent of full capacity with a vacuum of p^* permits the determination of the fuel input. With this fixed determination, any capability and fuel cost penalties resulting from failure to maintain a vacuum of p^* are computed as noted above.

SOME NOTES ON PROGRESS MADE

Our main effort in connection with the analysis of existing plants has gone into the development of a computer program to analyze alternative cooling system modifications. Our objective was to develop a program that was essentially a counterpart to the program we have completed for proposed plants.

In the appendix to this study paper we describe the existing plant computer program in its current form. Because we have not tested the program extensively we cannot be confident that it is free of the kind of programming "bugs" that are only revealed by testing. A central reason for not pursuing the testing stage in earnest is that we have become convinced that the portion of the program dealing with fuel cost and capability penalties must be revised. As indicated above we originally felt that these penalties would not play a dominant role in the analysis, but numerical results from the program indicated clearly that we were in error on this point.

Essentially, the nature of the revision that is required consists of introducing a seasonal analysis to more accurately account for the fuel and capability penalties. A good model for how this might be accomplished is contained in Fullers' work (Dynatach - July 8, 1970). In his study he essentially determined least cost cooling systems (with only passing regard for thermal quality constraints) on the basis of capital costs and operating costs; the latter were not determined on the basis of a seasonal analysis, but rather in a manner related to the approach taken in Appendix 4 herein. However, Fuller then subjected the so-called optimal design to a thorough seasonal analysis to obtain an improved estimate of the operating costs.

This latter aspect of Fuller's work is of central interest here. We would proposed that it be incorporated into our existing program as a sub-

routine to more accurately determine fuel and capability penalties on a seasonal basic for all designs considered. The least cost design to meet the thermal quality standards would then reflect all costs more accurately.

In the latter stages of the project we attempted to incorporate a seasonal analysis (as discussed above) into our own program. Unfortunately time and manpower constraints prevented us from getting past the stage of writeup specifications for the requisite subroutine. While we were unable to complete this portion of the work it did appear potentially promising (there were several iterative computations whose convergence properties had yet to be discussed). The main question however was whether it would have increased the computing time per design to the point where the large number of designs that need to be evaluated could have been examined in a reasonable time. Recall that it was the computational burden associated with the seasonal analysis that originally led us to attempt to avoid it.

There may be some who will argue that what we were attempting to do is merely computerize what power plant design engineers do regularly. However, this criticism misses two central themes which, to our knowledge, have not yet been developed. First we were attempting to develop an approach to determine the minimum cost combination of control options to meet a given thermal quality standard - the number of combinations of options we sought to examine was enormous in comparison with standard approaches. Second, and more importantly, we sought to use this methodology to develop functions giving cost as a function of the cooling system effluents' characteristics. The costs of attaining alternative levels of quality for our environment is the sort of information badly needed by our decision makers - but in very short supply.

APPENDIX

TECHNICAL SPECIFICATIONS FOR A COMPUTERIZED COOLING SYSTEM DESIGN PROCEDURE APPLICABLE TO EXISTING STEAM-ELECTRIC POWER PLANTS

1.0 INTRODUCTION

This appendix describes the assumptions, formulas and procedure embodied in a computer program developed to estimate costs of cooling system design modifications for existing steam-electric power plants, as discussed earlier in this study paper. The present appendix is similar both in format and content to Appendix A of CEM's final report, which deals with design and costing details for proposed generating stations. Thus, in the interest of brevity, extensive references to relevant sections of Appendix A will be made in the present appendix in many instances where the two program are identical.

Unlike the CEM cooling system design program for proposed plants which exists in operating form, the program for existing plant cooling system designs discussed herein is presently in unfinished and non-operational form.

As was the case of the proposed plant analysis, the cooling system design procedure for existing plants consists of the systematic determination of an array of alternative design variations and their associated cost characteristics for each of three, broadly defined, cooling system flow configurations, namely: once-through, recirculation, and helper. From the total set of desired design variations, the program then selects those subsets of designs with effluent temperatures and flowrates meeting a prescribed set of alternative thermal effluent standards. The minimum cost cooling system design meeting each such standard is then selected from each subset of technically feasible designs.

A preliminary assumption underlying our present approach is that the unit under consideration currently has a once-through cooling system. Given our overriding concern with thermal pollution abatement this represents the class of existing plants that is of primary interest. In addition, assume that the existing condenser will not be changed, and that existing pipes and pumps will be used to the extent possible. We assume here also that for any given plant analysis the firing capacity at the boiler is held constant. Thus, if a particular cooling system design modification involves an

increase in condenser backpressure compared with that of the existing once-through system, then the corresponding increase in station heat rate (decreased in thermal efficiency) will imply both an increase in unit fuel cost and a decrease in the unit's rated capacity. Penalty cost calculations of capacity and output associated with the existing plant on both capacity and energy accounts must thus be recognized. However, using a symmetric line of reasoning, if the design under consideration leads to a decrease in station heat rate, then we determine a corresponding capacity and fuel cost benefits (as opposed to penalty costs).

The computations described herein relate to modifications of an existing once-through cooling system that does not currently meet specified thermal quality standards. We consider the following possible modifications:

- Once-through, with and without blending—The modification of once-through cooling system effluents by altering the water flow rate through the condenser. Blending schemes consist of taking water (from the intake) directly to the cooling system outlet and mixing it with condenser outflow to decrease the effluent temperature.
- Recirculation—the formation of a closed loop for circulating water flow, with the addition of a heat rejecting device (e.g., cooling tower or pond) to remove the heat added to the circulating water flow at the condenser.
- Helper—the addition of a heat rejecting device to cool part or all of the outflow from the condenser prior to its discharge to a receiving water.

The computational procedure in its current form is restricted to "design conditions" involving 100 percent of capacity utilization. It would have to be modified to accommodate consideration of the production decrease option discussed in the body of this paper. In addition, preliminary numerical results using the procedure suggest that the penalty cost computations described below should be revised to account explicitly for seasonal variations in loading patterns and climatic conditions.

The remainder of this appendix is organized as follows. First, we describe the input data requirements and also some preliminary computations to yield characteristics of the existing cooling system. Then we outline the computational steps made in connection with each of the three types of cooling system modifications. Finally, we

outline the computations involved in a key subroutine used to determine station heat rates and condenser exit flow temperatures for alternative cooling systems.

2.0 INPUT DATA REQUIREMENTS

Before turning to the computational issues, we observe that the existing plant cooling system analysis requires nearly all the input data used for the proposed plant design (see Appendix A of the CEM final report document). In addition, the following characteristics specific to the existing plant are required as further input:

- Condenser area
- Design cooling water flow rate
- Design cooling water flow velocity
- Fraction of heat input lost from the boiler
- Points on the curve of thermal efficiency (heat rate) vs. steam condensing temperature for the turbogeneration.
- Plant thermal efficiency.*

Furthermore, it is necessary to specify an upper limit on the velocity of water flowing through the condenser, and the maximum permissible steam condensing temperature. As explained below, these two characteristics serve to constrain the set of feasible cooling system modifications.

3.0 ASPECTS OF THE EXISTING COOLING SYSTEM

In computing the costs and other characteristics for various modifications to the existing system, a need exists for information on several aspects of the existing cooling system. These characteristics are determined within the computer program in order to minimize the required input data and, more importantly, to guarantee that values of various cost elements computed for cooling system modifications are consistent with those computed for the original cooling system.

*This is subsequently revised in the manner described in Section 3 below, to guarantee that it is a value that is consistent with other input characteristics of the existing plant.

In particular, we compute the following values for the original cooling system:

- The head loss through the condenser; the computational procedure is given above in Appendix A, subsection 7.1 (iv). This loss is a function of the velocity of the circulating water, tube length, and the inside diameter of the tubes, all of which are taken as given (i.e., inputs to the computer program).

- Inlet and outlet pipe diameter computed as a function of flow rate and velocity (input variables) using the continuity equation of elementary mechanics.

- The head loss through the pipes from the intake supply to the condenser, and from the condenser to the effluent disposal system. This loss is computed using the Hazen-Williams equation which requires values of a smoothness coefficient, pipe length, flow rate, and pipe diameter; all these parameters are inputs except for the pipe diameter which is computed above.

- The pump horsepower required to overcome the head losses through the pipe and condenser (computed above) and the elevation head to be overcome in going from the intake supply to the condenser.*

- The annual operating cost for the existing once-through system; this includes the cost of water purchase (if any) and treatment (if any) plus the energy cost to run the pumps used in the circulating water system. The computation, which is essentially as described in subsection 7.5 of Appendix A, involves the following variables which are taken as given: percent of time the unit is available; unit costs of water purchase and treatment and auxiliary power. The pump horsepower, computed above, is also needed.

The only other preliminary computation that must be made concerns the thermal efficiency and condenser duty of the existing plant. The thermal efficiency plays a central role in the subsequent cost analyses. This results because the penalty costs associated with losses in capacity and power output in situations involving alternative cooling system designs are based, in part, on the differences in efficiency in the cases involving alternative designs and the case of the existing plant. For this reason it is essential that the procedure for computing the efficiency of the existing plant be consistent with the procedure used for situations involving alternative cooling system designs.

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*The requisite formulas for the various head loss computations are given in Appendix A, subsection 7.4.

The computation of thermal efficiency for existing system is, however, a non-trivial matter since efficiency is linked directly to the condenser backpressure, and consequently the overall performance of the existing once-through cooling system. The iterative procedure used to make this computation is described in detail in Section 7 below. The procedure requires that the following parameter be specified: condenser area and intake water temperature, flow rate and velocity; and a trial value of thermal efficiency. These parameters are among the input data requirements specified in Section 2 above.

The computational procedure of Section 7 also requires a trial value of condenser duty $\hat{H}_o$, which is computed using

$$\hat{H}_o = 3413\ P_o \left(\frac{1}{\hat{\eta}_o} - 1 - \frac{s}{\hat{\eta}_o} \right)$$

where the variables on the right-hand side are inputs and are as described below:

P_o = rated existing plant capacity (kw)

$\hat{\eta}_o$ = trial thermal efficiency of existing plant (%/100)

s = fraction of heat rejected to atmosphere (via stack) (%/100)

This relationship may be deduced by observing that the number of Btu's required per kw hr of electricity produced is, by definition of power plant thermal efficiency, equal to $3413/\hat{\eta}_o$. Of this 3413 Btu is actually used to produce a kw hr of electricity, a fraction, s (approximately .1 for fossil fuel plants and zero for nuclear plants) is lost through inefficiency in combustion, and the remainder, namely,

$$\frac{3413}{\hat{\eta}_o} - 3413 - s \left(\frac{3413}{\hat{\eta}_o} \right)$$

is waste heat transferred at the condenser (i.e., condenser duty). This relationship holds for 1 kw hr of electrical energy produced. For a unit producing P_o kw hr/hr, the duty is given by Eq. (1) above.

4.0 ONCE-THROUGH, WITH-AND-WITHOUT BLENDING

In the once-through systems without blending the existing system stays "as is" to a large extent. (The pipe layout diagram given in Section 3 of Appendix A applies in this case.) The only variations considered are changes in the rate of flow of the circulating water. For flow rates greater than the design flow rate of the existing cooling system, capital costs for increased pumping capacity will be incurred. Additional operating costs (or benefits) include the following:

- Penalty costs (or benefits) due to changes in capability and fuel consumption that result from shifts in net station heat rate; such shifts are caused by the impact of variations in flow rate on the condenser vacuum.
- Costs resulting from the changed flow rate and head loss conditions.
- Costs for water purchase and treatment, inasmuch as these costs (where applicable) are directly related to flow rate.
- Maintenance costs associated with added pumping facilities.

For each once-through system examined, we also consider the possibility of blending at various levels—i.e., taking water from the intake directly to the cooling system outlet and mixing it with the condenser outflow to decrease the effluent temperature.

Figure A1 is a flow chart of the computational sequence for alternative once-through designs. The various steps in this procedure are discussed below in turn.

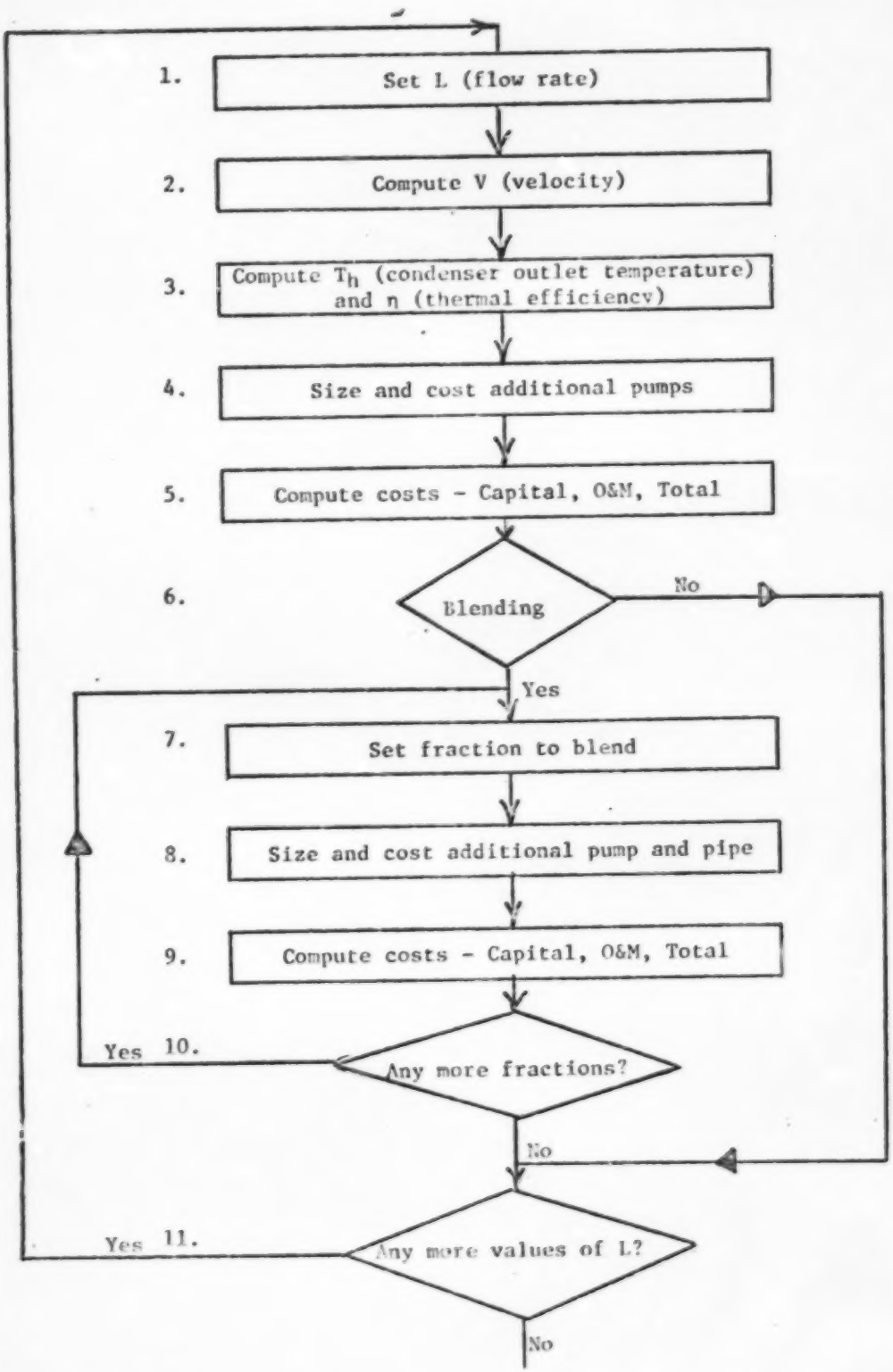
Step 1: Set circulating water flow rate (L)

Before establishing a value for L, the flow rate of water through the condenser, it is necessary to determine the maximum and minimum values that it can attain. The maximum possible value of L is taken as the lowest of the following three values:

- The maximum flow rate available from the source of intake supply, $\bar{L}$.
- The flow rate, associated with the maximum feasible velocity, $\bar{V}$, that can be accommodated by the existing condenser and intake pipes. Using the continuity equation of elementary fluid mechanics, and recognizing that cross sectional area is constant since the existing pipes are to be used, this flow rate value is computed as

FIGURE A1

FLOW CHART FOR ONCE-THROUGH SYSTEM DESIGN COMPUTATION



$$\bar{V} \frac{L_o}{V_o}$$

where L_o and V_o, the flow rate and velocity associated with the existing system, are input to the computer program. Observe that $\bar{V}$, also an input variable, will generally be about 8—10 ft/sec; this is considered to be a working upper limit for velocities through steam surface condenser used in conventional power plants.*

• The flow rate associated with R_{min} (F°), the minimum feasible temperature increases across the condenser, which is taken as an input parameter. Using a basic relationship from elementary thermodynamics†, this flow rate is computed as $H_o \div 500\ R_{min}$, where H_o (Btu/hr) is the condenser duty associated with the existing plant. This represents an approximate bound on flow rate inasmuch as H will not be the precise value of the condenser duty when modifications in the existing design are made; however, the approximation is quite satisfactory for purposes of establishing a bound.

The minimum possible value of flow rate is established by observing that individual turbines have a maximum backpressure beyond which they cease to operate—a result of automatic shutdown. This backpressure establishes an upper limit on steam condensing temperature which we take as a given value. Since the practice in steam surface condenser design is to provide at least a 5°F temperature difference between the temperature of the condensing steam and T_h (the temperature of circulating water leaving the condenser), the limit on steam condensing temperature establishes an upper limit on T_h which we refer to as T_{hmax}. Furthermore, since the temperature of the circulating water entering the condenser is fixed at T_a, the design ambient intake water temperature, the approximate minimum flow rate may be determined as

$$\frac{H_o}{500\ (T_{hmax} - T_a)}$$

*Newbold, H. L. R., Jackson & Moreland, Boston, Mass., personal communication, August 12, 1970.

†See McCoy, [12, p. 22].

The values of L to be considered in defining alternative once-through designs are set recursively starting from the upper bound computer above, and decreasing in specified constant increments of flow rate to a minimum value which is at least as great as the lower bound computed above. For each value of L there is an associated once-through design and, possibly, several other designs characterized by different levels of blending.

Step 2: Compute velocity of circulating water flow (V)

Since the intake and outlet piping in the cooling system is fixed in cross sectional area (the piping is part of the existing system), the continuity equation can be used directly to determine the velocity (V) associated with the flow rate (L). In particular, the velocity is given by

$$V = L \frac{V_o}{L_o}$$

Step 3: Compute condenser outlet temperature (T_h) and plant thermal efficiency (η)

The computation of T_h and η (and incidentally H, the condenser duty) involves an iterative procedure that is described in detail in Section 7, below. The main variables needed to initiate this iterative procedure are as follows:

T_a = temperature of water entering the condenser

L = flow rate through the condenser

V = velocity of water flow through the condenser

A_o = cross sectional area of condenser

$Tcon_o$ = design steam condensing temperature for existing system

η_o = design plant thermal efficiency for existing system

H_o = design condenser duty for existing system

The fact that the velocity and flow rate through the condenser are different from those in the existing cooling system suggests that the condenser duty, temperature of water leaving the condenser, plant thermal efficiency, and temperature of the condensing steam will also be different. The iterative procedure mentioned above is designed to compute these new values of condenser duty, etc.

Step 4: Size and cost additional pumps

In Section 3 above, we indicated that as a preliminary step we compute (for the existing system) the pump horsepower required to overcome the head losses through the pipes and condenser, and the elevated head. For convenience, we refer to that pump horsepower requirement as H_{po}.

At this point in the program the same standard procedures are used to compute the head losses through the pipes and condenser. It is necessary to compute new values for these losses since they are a function of velocity; and V, the velocity established in Step 2, will generally not be equal to V_o, the velocity in the existing cooling system.

A new pump horsepower requirement, H_p, can be computed and compared to H_{po}, the pump horsepower provided in the original system. If $H_p > H_{po}$, it will be necessary to purchase additional pumping equipment to make up the difference in horsepower. The procedures used to estimate the cost of the additional pumps are identical to those discussed in subsection 7.4 of Appendix A.

Step 5: Compute costs: capital, O&M, total

Here we compute the additional annual cost that would be incurred if the existing cooling system were modified by the addition of new pumps. Following a standard procedure, the equivalent annual cost is determined by multiplying the capital cost for new pumps by the fixed charge rate.

The O&M costs include costs for water purchase, if any, and treatment, and maintenance costs associated with the pumps. These costs are computed using standard procedures (see subsection 7.5 of Appendix A), and then they are summed. The additional O&M cost is taken as the difference between this sum and the O&M costs of the existing cooling system (the computation of the latter was discussed in Section 3 above).

There are still additional costs of some consequence, namely, the penalty associated with the capacity loss resulting from the use of extra pumps and with modified plant efficiency; and also the cost to generate power from an alternative source to compensate for deficiencies in power output occasioned by the loss in generating capacity caused by decreased plant efficiency.*

*As explained below, in the event that the thermal efficiency associated with the cooling system design under consideration is increased relative to the original design, it will be necessary to account for a negative cost (i.e., a benefit).

We account for the capacity loss occasioned by the use of extra pumps by determining the following penalty cost:

$$0.746 \Delta H_p \gamma_2$$

where ΔH_p is the added pump horsepower, γ_2 is a penalty cost for capability loss in \$/kw-year, and 0.746 converts horsepower to kilowatts.

Accounting for the capacity change occasioned by the change in plant thermal efficiency is somewhat more complicated in the sense that it requires a preliminary discussion of how changes in thermal efficiency may be translated into alterations in capacity. Observe that, by definition, plant thermal efficiency is the ratio of electricity output to heat input. Thus, for an original firing rate of F_o Btu/hour and an original design output of P_o kwhr/hr, the plant thermal efficiency at design capacity is

$$\eta_o = \frac{P_o}{(F_o \div 3413)}$$

where 3413 is a factor representing the number of Btu's in one kilowatt hour.

Observe further that as the original cooling system is modified (in once-through designs by means of varying the water flow rate) the steam condensing temperature will be altered. This temperature, as well as the thermal efficiency associated with modified cooling system designs, is determined in Step 5 above. Consequently, at this step in the computation, η, the thermal efficiency associated with the cooling system design under consideration, is a known quantity.

For purposes of computing P, the plant capacity associated with the efficiency η, we postulate that the firing rate will be held constant at F_o Btu/kwhr.* Thus, it follows from the definition of thermal efficiency that the capacity associated with η will be

$$P = \frac{\eta F_o}{3413}$$

*This assumption is also made in the iterative procedure for computing η (see Section 7).

Straightforward algabraic manipulation of the two preceding equations leads to the following relationship for the change in capacity ($P_o - P$) as a function of the original capacity, P_o, the original thermal efficiency, η_o, and the thermal efficiency associated with the design under consideration, η; i.e.,

$$P_o - P = P_o \left(\frac{\eta_o - \eta}{\eta_o} \right)$$

All quantities on the right-hand side are known at this state of the computation.

Finally, the annual cost attributable to this plant capacity (or capability) change is computed as

$$0.746\ P_o \left(\frac{\eta_o - \eta}{\eta_o} \right) \gamma_2,$$

which has an interpretation similar to the cost for capacity loss associated with the additional pumping facilities. Observe that for cases where increased water flow rates lead to increased efficiencies, the expression above yields a negative overall cost (i.e., a benefit), as it should since capacity will be effectively increased as a consequence of the improved efficiency.

In addition, account must be taken of the additional energy (fuel) cost of generation occasioned by the change in thermal efficiency.* (If the thermal efficiency increases, the following generating cost penalty yields a negative result, i.e., a benefit.) Assuming the plant operates with the same load factor of F_1, the change in power output is

$$8760\ F_1\ P_o \left(\frac{\eta_o - \eta}{\eta_o} \right) \text{ kwhr/year.}$$

This quantity multiplied by γ_1 \$/kwhr (the unit cost of generating power from an alternative source) yields the additional annual fuel cost (or benefit) resulting from the change in thermal efficiency.

*As noted earlier in the body of this study paper, we are dissatisfied with this approach to penalty costs, and suggest that a seasonal analysis must be employed to develop a more appropriate procedure.

Steps:

6. Blending?
7. Set fraction to blend.
8. Size and cost additional pump and pipe.
9. Compute costs.
10. Any more fractions?
11. Any more values of L?

Steps 6—11 above correspond to Steps 6—11 in the design procedure for once-through systems in the proposed plant analysis (Section 3 of Appendix A).*

5.0 RECIRCULATION SYSTEMS

In this section we consider the modifications to the existing once-through system required to convert it to a recirculating system. The modifications are developed in keeping with the assumptions regarding the desirability of minimizing the time required to make the modification and the desirability of utilizing facilities and equipment already in place. Consequently the changes involve the installation of a heat rejection device and the requisite piping and pumping facilities to complete a closed loop.

Figure 2 shows the pipe layout for the recirculating system. The solid lines represent components already in place (i.e., the existing once-through system); these include the condenser, inlet and outlet piping, and the inlet pumping facilities. The dotted lines represent cooling system components that must be added in order to convert the existing once-through system to a recirculation system; these include new pipes and pumps and a heat rejection device (i.e., a mechanical or natural draft cooling tower to a cooling pond).

It is assumed that the added pipe will be of the same diameter as the existing pipe to yield a constant velocity for the recirculating water. Cooling systems are often designed to maintain a constant velocity throughout; the assumption that all pipe diameters are identical is consistent with this. It is also assumed that the existing pumping facilities will be salvaged to take up the relatively low pumping requirements for

*Note however that Step 11 in the proposed plant analysis refers to T_h instead of L, so Step 11 is not quite identical in the two cases.

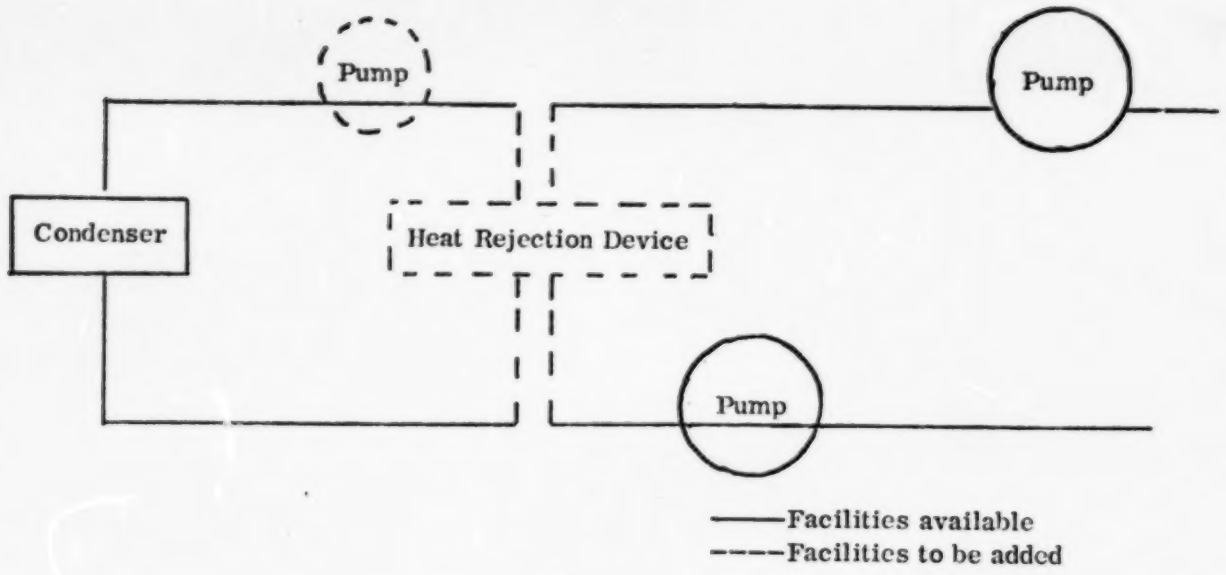


FIGURE A2

PIPE LAYOUT FOR RECIRCULATION SYSTEMS

make-up and blow-down flows. Consequently, the only new pumping facilities required will be those necessary for the recirculating water flow.

The computations involved in designing and costing out modifications to the existing system are outlined in Figure A3 and described individually below.

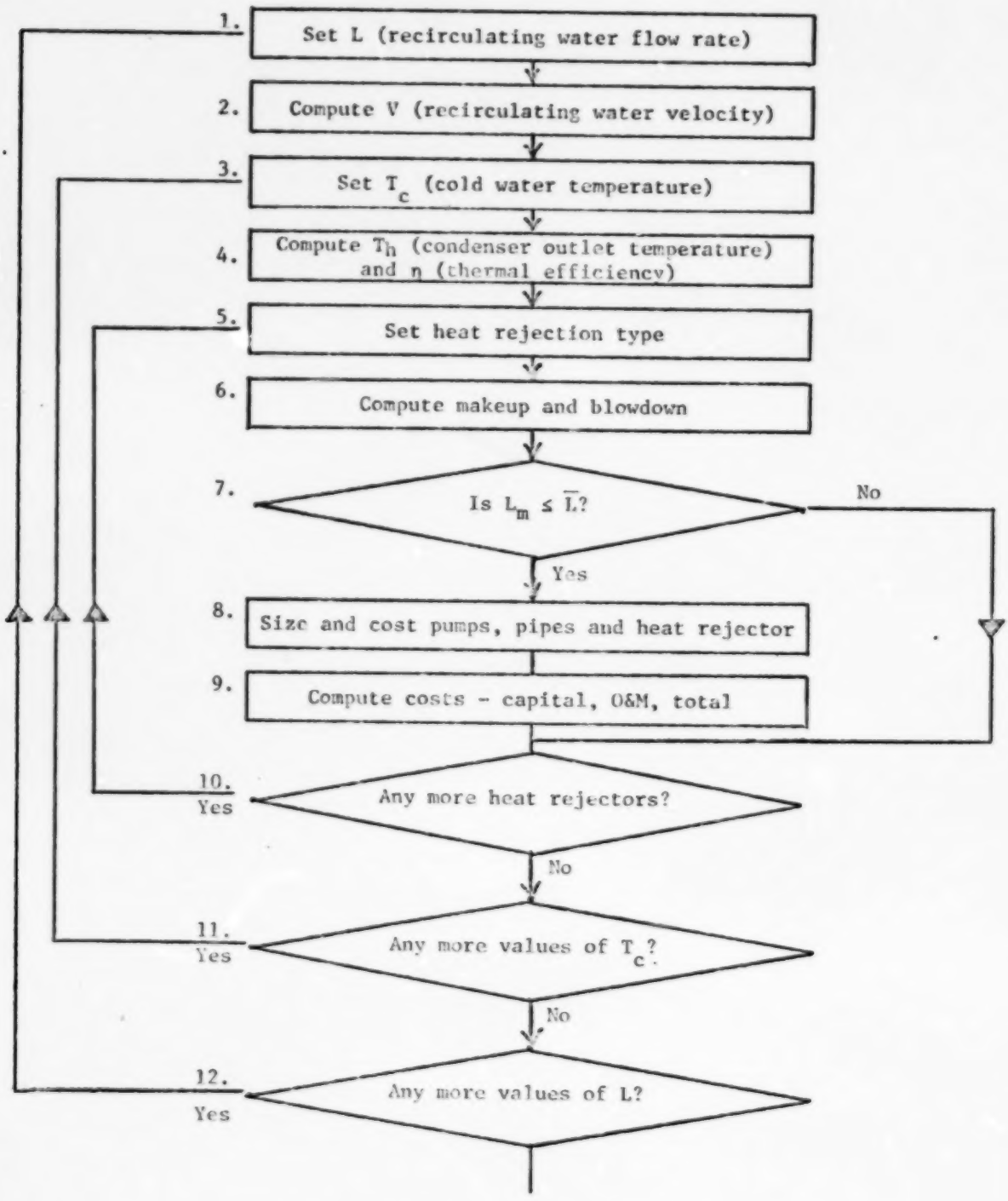
Step 1: Set recirculating water flow rate (L)

Before establishing a value for L, the flow rate of water through the condenser, it is necessary to determine the maximum and minimum values that it can attain. The maximum possible value of L is taken as the flow rate associated with the maximum feasible velocity (i.e., as per Step 1 of the once-through system design computations).

The minimum possible value of flow rate can be determined by observing that the maximum possible temperature rise across the condenser is fixed by physical considerations. In particular, the maximum possible temperature of water leaving the condenser, as described in Step 1 of the once-through design, is T_{hmax}. The minimum possible temperature of water entering the condenser (T_{cmin}) is equal to the sum of

FIGURE A3

FLOW DIAGRAM FOR RECIRCULATING SYSTEM DESIGN COMPUTATIONS



the ambient wet bulb temperature and the minimum feasible "approach" (this will only be approximate for cooling ponds).* Consequently, an approximate minimum flow rate may be determined as

$$\frac{H_o}{500\ (T_{hmax} - T_{cmin})}$$

where H_o is the condenser duty associated with the existing plant.

The values of L to be considered are set recursively starting from the upper bound computed above, and decreasing in specified constant increments of flow rate to a minimum value which is at least as great as the lower bound computed above. For each value of L there will be a number of recirculating system designs corresponding to alternative values of water temperature entering the condenser and heat rejection types.

Step 2: Compute velocity of recirculating water flow (V)

Since the piping in the recirculating flow loop is of constant diameter, the velocity can be computed as a function of flow rate using elementary principles of fluid mechanics (i.e., as per Step 2 of the once-through design procedure).

Step 3: Set cold water temperature (T_c)

As in the case of flow rate, the values for the temperature of the cold water entering the condenser are set recursively starting from an upper bound, and decreasing in specified constant increments to a minimum value which is greater than a physically established lower bound.

The procedure for establishing a lower bound, T_{cmin}, was noted in Step 1 above. Below we consider the establishment of the upper bound, T_{cmax}, which will turn out to be a function of the circulating water flow rate.

As mentioned in the discussion of Step 1 above, there is a maximum possible temperature for water leaving the condenser, T_{hmax}. (Recall that this is established by the fact that turbines are built with a maximum allowable backpressure beyond

*This constraint does not apply to cooling ponds; but we choose to use it rather than revise the procedure to treat cooling ponds in a set of separate, but nearly identical, computations.

which automatic shutdown takes place.) The temperature increase across the condenser is given approximately by

$$\frac{H_o}{500\ L}$$

where this value is approximate because H_o, the original condenser duty will not be precisely equal to the condenser duty with alternative cooling system designs. Thus, the approximate maximum temperature of water entering the condenser is given as

$$T_{cmax} = T_{hmax} - \frac{H_o}{500\ L}$$

where this value is approximate because H_o, the original condenser duty will not be precisely equal to the condenser duty with alternative cooling system designs. Thus, the approximate maximum temperature of water entering the condenser is given as

$$T_{cmax} = T_{hmax} - \frac{H_o}{500\ L}$$

Steps:

4. Compute condenser outlet temperature (T_h) and plant thermal efficiency (η)

These computations involve an iterative procedure which is described in detail in Section 7 below.

5. Set heat rejector type

6. Compute makeup and blowdown

7. Is makeup water flow (I_m) ≤ maximum feasible intake flow ($\bar{I}$)?

Steps 5—7 above are identical to Steps 5—7 in the design procedure for recirculation systems in the proposed plant analysis (Section 4 of Appendix A).

8. Size and cost additional pumps, pipes, and heat rejector.

The relevant formulas and assumptions in carrying out these computations are discussed in subsections 7.2 through 7.4 of Appendix A. The main assumptions relating to this particular step were mentioned above but are repeated here for completeness. That is:

(1) the added pipe will be of the same diameter as the existing pipe, and

(2) the only additional pumping facilities needed are for the recirculating flow loop.

9. Compute costs: capital, O& M, total

Here we compute the additional annual cost that would be incurred if the existing cooling system were modified to a recirculating system with a recirculating flow rate of L, condenser inlet temperature of T_c, and heat rejector of a fixed type. The capital costs include those for additional pipes, pumps and heat rejector. The equivalent annual cost is determined by multiplying the capital cost by the fixed charge rate.

The O& M costs as well as penalty costs for extra pumps and changes in thermal efficiency are determined in much the same manner as described in Step 5 of the procedure for alternative once-through modifications of the existing system (i.e., Section 4 above).

10. Any more heat rejectors?

11. Any more values of T_c?

12. Any more values of L?

Steps 10--12 above are identical to Steps 10—12 in the design procedure for recirculating systems in the proposed plant analysis (Section 4 of Appendix A).

6.0 HELPER SYSTEMS

The modification of the existing once-through system to a helper system involves the addition of a heat rejecting device (i.e., a mechanical or natural draft tower or cooling pond) to treat all or part of the flow leaving the condenser before it reaches the effluent disposal system. The modifications consist of the addition of a heat rejection device with the associated piping and pumping facilities. These additional facilities are shown as dotted lines in Figure 4, which is a diagram of the system layout.

It is assumed that the existing pipe will be used to the extent possible, the only new pipes being those required to transport the flow when it is divided with a portion going to the heat rejection device. Furthermore, it is assumed that the existing pumps at the source of intake water will be augmented as necessary to meet any increased requirements, and that a completely new pumping facility will be added in the pipeline just before the heat rejection device.

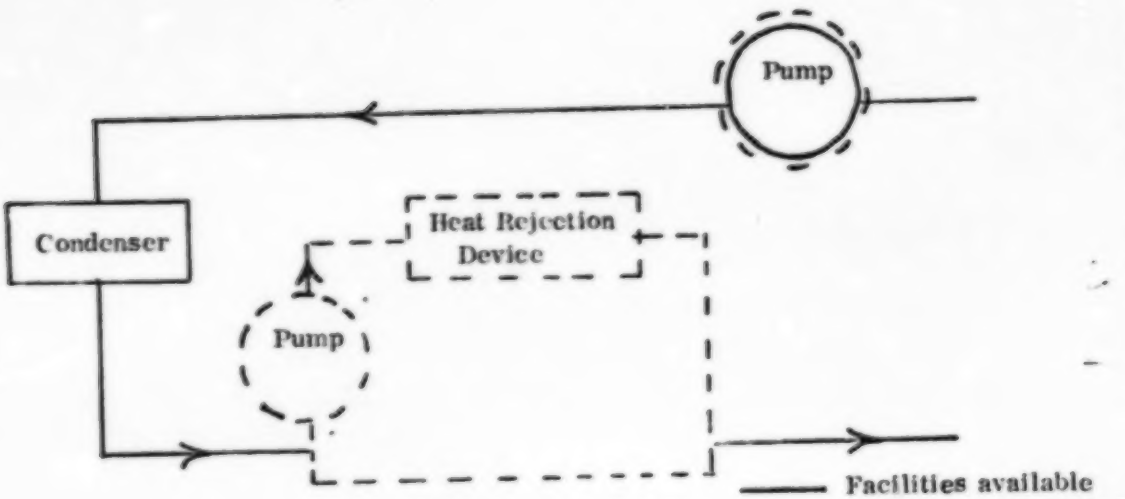


FIGURE A4

PIPE LAYOUT FOR HELPER SYSTEMS

The computations involved in designing and costing out modifications to the existing system are outlined in Figure A5 as described below.

Steps:

1. Set intake water flow rate (L) and compute velocity of circulating water flow (V)
2. Compute condenser outlet temperature and plant thermal efficiency

Steps 1 and 2 are carried out in the same manner as the first three steps in the design procedure for once-through-with-blending schemes for existing plants (see Section 3 above).

3. Set temperature of flow leaving heat rejection unit (T_c)
4. Set heat rejector type
5. Set π, the fraction of flow leaving the condenser that is diverted to the heat rejection unit
6. Compute flow rate and heat load to heat rejection unit
7. Compute effluent flow rate and temperature

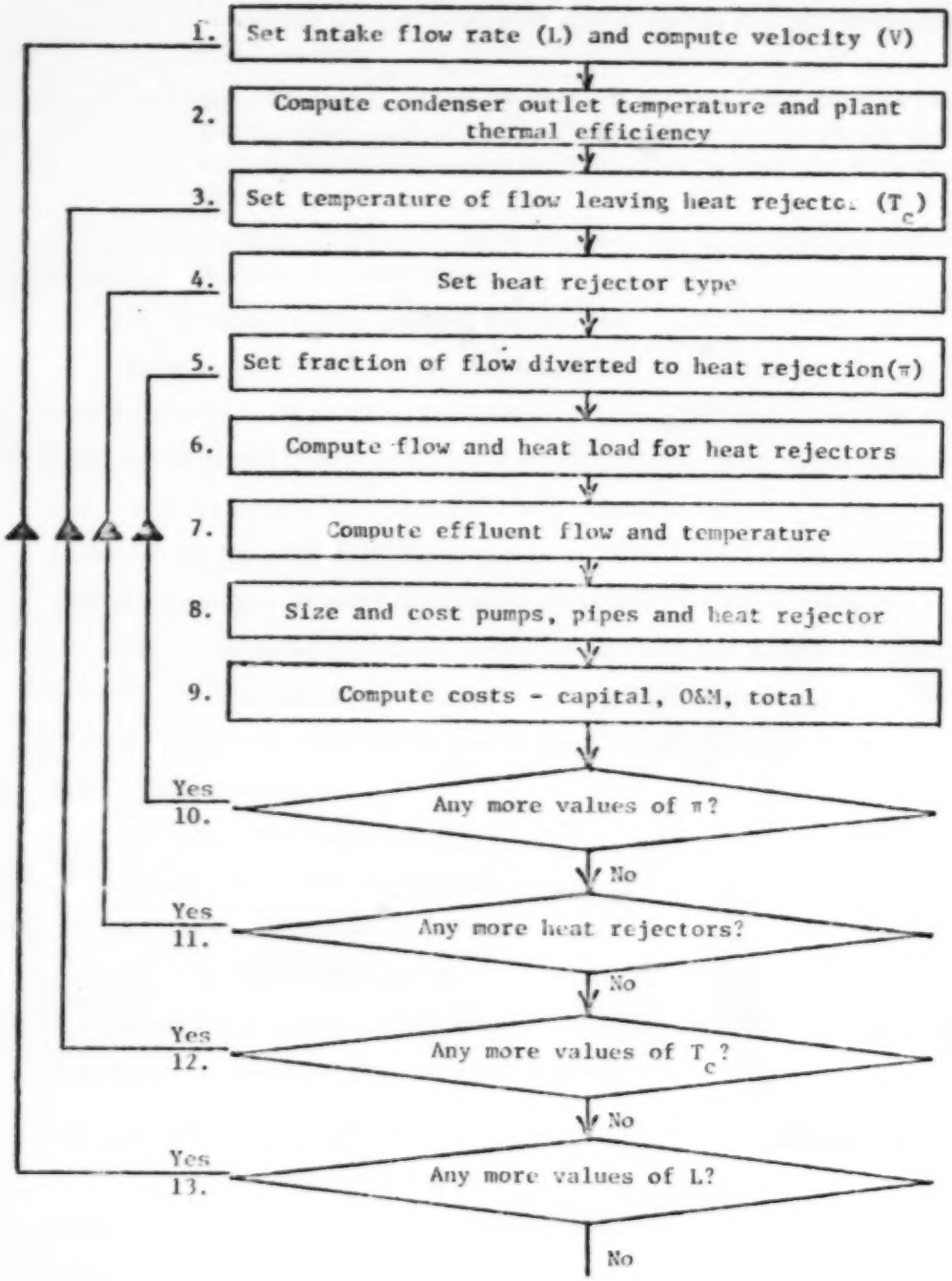
Steps 3—7 above are identical to Steps 3—7 in the design procedure for helper systems in the proposed plant analysis (Section 5 of Appendix A).

8. Size and cost additional pumps and pipes and heat rejector

The relevant formulas and assumptions in carrying out these computations are discussed in subsections 7.2 through 7.4 of Appendix A. The main assumptions relating to this step were mentioned above, but are repeated here for completeness.

FIGURE A5

FLOW CHART FOR HELPER SYSTEM DESIGN COMPUTATIONS



They are:

(1) The only new pipes are those required to transport the cooling water flow when it is divided with a portion diverted to the heat rejection unit.

(2) Additions to the intake pumping facilities are made as needed. The approach here is similar to that described for Steps 4 of the once-through design procedure for existing plants in Section 4 above.

(3) New pumping facilities are required in the pipeline just before the heat rejection device.

9. Compute costs: capital, O & M, total

Here we compute the additional annual cost that would be incurred if the existing cooling system were modified to a helper system with a heat rejection device of a given type, and an intake flow rate of L, a flow diversion to the heat rejection unit of π L, and a temperature of flow leaving the heat rejection unit of T_c. The capital costs include those for additional pipes, pumps and heat rejector. The equivalent annual cost is determined by multiplying the capital cost by the fixed charge rate.

The O & M costs as well as penalty costs for extra pumps and changes in thermal efficiency are determined in much the same manner as described in Step 5 of the procedure for alternative once-through modifications of the existing system (i.e., Section 4 above).

10. Any more values of π?

11. Any more heat rejectors?

12. Any more values of T_c?

13. Any more values of L?

Steps 10—13 above are identical to Steps 10—13 in the design procedure for helper systems in the proposed plant analysis (Section 5 of Appendix A).

7.0 COMPUTATION OF THERMAL EFFICIENCY AND CONDENSER DUTY

At various points in this appendix reference has been made to an iterative procedure for determining thermal efficiency and condenser duty. This procedure, which takes the form of a subroutine in the computer program, is described below.

The following variables are known at the time the procedure is initiated:

A_{so} = surface area of condenser;

T_c = temperature of water entering the condenser;

L = flow rate of water entering the condenser;

V = velocity of water entering the condenser;

s = fraction of heat input lost out the stack;

P_o = design capacity of the existing plant;

H_o = condenser duty of the existing plant; and

η_o = thermal efficiency of the existing plant.*

The output from this iterative procedure consists of a set of consistent values of the following variables: condenser duty, thermal efficiency, steam condensing temperature, and temperature of the water leaving the condenser.

There are two computations that are made prior to the initiation of the iterative procedure. First, it is necessary to compute the value of u, the heat transfer coefficient for the condenser. This is done using the equations presented in subsection 7.1 (i) of Appendix A1 these equations permit the determination of u as a function of velocity, V, and condenser inlet temperature, T_c. Second, it is necessary to establish a trial value for condenser duty. For this purpose we set $H' = H_o$ where H' is the trial value, and H_o is the condenser duty for the existing plant.

The iterative procedure consists of the following six steps:

1. Compute T_h, the temperature of water leaving the condenser, using

$$T_h = T_c + \frac{H'}{500\ L} .$$

This is a basic equation from elementary thermodynamics [2, p. 32].

2. Compute ΔT_m, the log-mean-temperature-difference, using

$$\Delta T_m = \frac{H'}{A_{so} u} .$$

This also is a standard relationship from thermodynamics [2, p. 349].

*When the procedure of this subsection is used to compute H_o and η_o, substitute $\hat{H}_o$ for H_o and $\hat{\eta}_o$ for η_o (see Section 3 above).

3. Compute T_{con}, the temperature of the condensing steam, using

$$T_{con} = \frac{T_c - e^{\varphi} T_h}{1 - e^{\varphi}} ,$$

where φ is defined by

$$\varphi = \frac{T_h - T_c}{\Delta T_m}$$

This relationships follows from the defining equation for ΔT_m (see Section 7.1 (i) of Appendix A), namely,

$$\Delta T_m = \left(\frac{T_h - T_c}{\ln \frac{T_{con} - T_c}{T_{con} - T_h}} \right)$$

Rearranging and utilizing the definition of φ, gives

$$\frac{T_h - T_c}{\Delta T_m} = \varphi = \ln \left(\frac{T_{con} - T_c}{T_{con} - T_h} \right)$$

Taking anti-logs,

$$e^{\varphi} = \frac{T_{con} - T_c}{T_{con} - T_h}$$

Finally, solving for T_{con} gives the relationship used above.

4. Compute η, the plant thermal efficiency, as

$$\eta = e_o + e_1 T_{con} + e_2 T_{con}^2$$

where e_o, e_1 and e_2 are known constants which are determined from a quadratic "least-square" curve fit of points on a curve of plant thermal efficiency vs. steam condensing temperature. This curve is developed from information supplied by the turbine manufacturers for the particular turbine-used.*

*Such curves are generally given in terms of condenser backpressure as opposed to the temperature of the condensing steam. However, the transformation from one variable to the other can be made using standard tables [1].

$$F_o = \frac{\eta_o}{3413\ P_o} = \frac{\eta}{3413\ P} ,$$

or

$$P = \frac{\eta}{\eta_o} P_o .$$

Substituting this equality into the above equation for H" yields

$$H'' = 3413 \left[\frac{1}{\eta} - 1 - \frac{s}{\eta} \right] \frac{\eta}{\eta_o} P_o ,$$

as required.

6. Compare the value of condenser duty computed in Step 5 above, H", with the trial value, H'. If these values are within a specified "error tollerance" then the iterative procedure is complete. Otherwise, H" is taken as the new trial value and Steps 1—6 are repeated. An input parameter controlling the maximum number of iterations is used to prevent the occurrence of an unbounded number of iterations.*

8.0 REFERENCES

1. Heat Exchange Institute, Standards for Steam Surface Condensers, 6th ed. (Heat Exch. Inst., N.Y., N.Y., 1970).

2. Mooney, D. A., Introduction to Thermodynamics and Heat Transfer (Prentice-Hall, Inc., Englewood Cliffs, N.J., 1955).

*We have not performed a mathematical analysis to determine the convergence properties of the iterative scheme; however, preliminary numerical computations showed convergence in less than 25 iterations.